Robotics Research Technical Report

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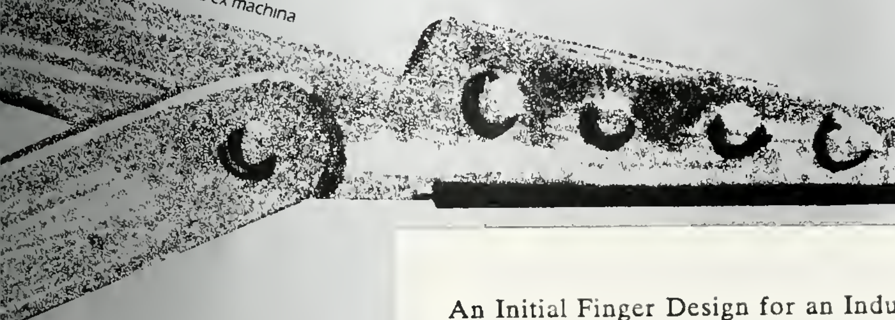

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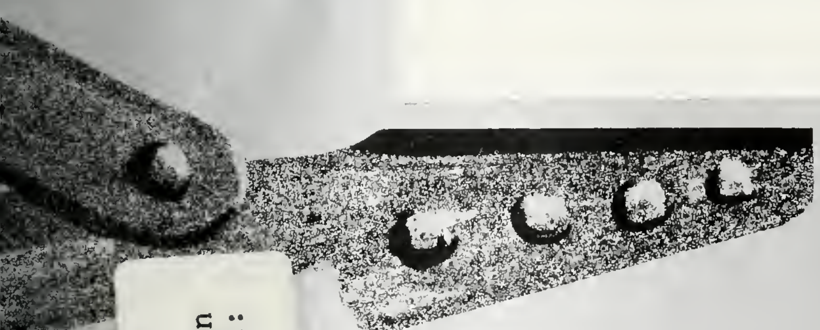
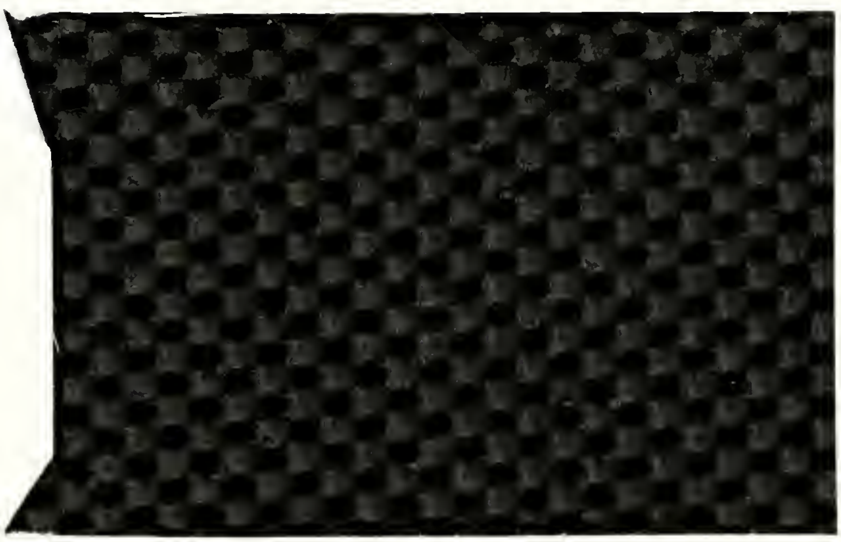


New York University
Cc nstitute of Mathematical Sciences

Computer Science Division
.51 Mercer Street New York, N.Y. 10012



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An Initial Finger Design for an Industrial Hand: Part II*

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1. Introduction

In the technical report by Walker (1985) a finger design was described which had two finger joints that were actuated by a single pneumatic spring return piston at each joint. When this finger was built and operated it was found that the small pistons were powerful enough to actuate the finger and that the actuation scheme, which exploited the static pressure associated with a steady flow compressed air line, allowed the piston to be exposed to pressures that ranged from atmospheric to the full reservoir pressure, by merely varying the degree to which a ball valve in the main compressed air line was closed. These tests subsequently showed that the actuation scheme and mechanical design of the finger were sound and as a result it was decided to design and build a more advanced finger that would have two additional degrees of freedom and a control system that would allow the position and compliance of all its joints to be controlled automatically. This report will therefore describe these changes in design and address the issues of instrumentation and control that are associated with the new finger design.

2. Mechanical Design

The new finger design (Fig. 1) was based on the initial finger design to the extent that the first two sections (i.e. the two parts that are connected at joint 1 in Fig. 1) of the finger were similar in shape to the equivalent parts in the original design thereby maintaining the same working envelope for joints 1 and 2 as existed in the initial design. The third part of the finger, however, differed from the initial design in that it was no longer rigidly attached to a fixture but was instead attached to a fixture *via* a universal type of joint (i.e. joints 3 and 4 in Fig. 1). This increase in the number of joints, from 2 to 4, not only increased the finger's working envelope in the $x-y$ plane (Fig. 2), which was a result of solely adding joint 3, but, also allowed the finger to move in the z direction (in an arc of approximately $\pm 30^{o}$) by rotating about joint 4. In addition, in an effort to add compliance control and to provide better position control, a second piston was added to each joint in the new design. The two pistons were then arranged (Fig. 3) in such a manner that each would exactly oppose the

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motion of the other. In this way control over the finger's position could then be achievable by creating a pressure difference between the two pistons whilst control over the finger's compliance could be achievable by varying the absolute pressure equally in both pistons. For example, in Fig. 3 the finger tip could be rotated anti-clockwise, about bearing D, by increasing the pressure in cylinder 2 whilst keeping the pressure constant in cylinder 1: on then achieving the desired position the finger's compliance could then be increased by increasing the pressure in cylinder 1 to that of cylinder 2.

3. Instrumentation

Unlike the original finger, which was not instrumented, the new finger design required instrumentation that would allow the position and compliance of each joint to be controlled automatically. In the case of the position control the most direct feedback parameter would have been joint angle, however, since angular displacement transducers tend to be larger than linear displacement transducers it was decided to measure the displacement of the piston that produces the joint rotation rather than the joint angle itself in order to minimise the size of the finger; this was done by mounting an LVDT directly on top of the piston (Fig. 4). However, the saving in physical size was paid for in terms of an increase in the complexity of the control loop since the relationship between the displacement of the piston and the joint angle was not linear. In the following section, however, it will be shown that this increase in control complexity was not too sever thereby justifying the decision to opt for a saving in finger mass over control loop simplification. At a later stage of design it might also be advantageous to attempt to construct a devise that incorporated the LVDT as part of the piston body.

In the case of the compliance control the desired feedback parameter was the piston pressure, which could be obtained by measuring the pressure in the lines that feed each piston. This could be done at a point close to the base of the finger (in a complete hand this could be done in the palm of the hand) thus allowing the pressure to be measured at a point close to the pistons in question thereby keeping the velocity-time delay to as small a value as possible, whilst at the same time allowing the transducers to be mounted at a point where they would not increase the physical size of the finger. An additional saving in size could also be achieved by using a single pressure transducer and a scanning valve (i.e. a pneumatic multiplexer), which could monitor all the pressure lines sequentially. It should be noted that despite the fact that the pressure in the piston would be different than that in the line no advantage would be gained, in terms of measurement accuracy, by measuring the pressure in

the piston since any transducer would have to be attached to the piston via a tube which was at least as small as the tube that feed the piston: the only difference between the two measuring schemes would then be that the line pressure measurement would lead the actual piston pressure whilst the piston pressure measurement would lag the actual piston pressure.

Finally, a tactile sensor would be required to detect touch and/or slip in order to provide some information about the condition of any grip that the finger tried to exert on a body (different types of tactile sensors are discussed by Patterson and Nevill (1985) and Walker (1986)). Initially this need only be a simple switch to indicate that contact had been made with a body whilst the more advanced task dependent tactile sensors would only be required at a lated stage of development.

4. Control Schemes

The pistons that were used to actuate the finger joints were available with and without a spring return and as a consequence a conventional control loop was designed around each of these two types of piston in order to ascertain which piston type would allow the optimum control of the finger to be achieved. In addition, in order to provide further control options, an unconventional bang-bang type of control scheme was also designed for both of the above cases. In order therefore to allow a choice to be made between these four control options it was necessary to obtain information about each schemes response characteristics, which was done by examining in detail each scheme individually. The results of these examinations are presented in the following sections and involved an analysis that assumed, Coulomb friction and an instantaneous piston pressure responded to any changes that occurred at the control valve.

4.1. Conventional Type of Control

In this report the term conventional control is used to imply a control system that feeds the error, between the desired and actual value of the quantity to be controlled, directly back into the control loop (Fig. 5) and in the case of this finger the quantity to be controlled was the finger position.

4.1.1. System using a Piston without a Return Spring

When the system that employed a piston without a return spring was examined, it was found that the piston's acceleration was purely a function of the pressure difference (p) between the two pistons and the friction (F_f) resisting the motion of the pistons (Eqn. 1).

$$m\frac{d^2x_0}{dt^2} = pA - F_f \tag{1}$$

On a further examination of this equation it was noted that automatic position control, of this system, was achievable by allowing the pressure difference between the two pistons to be proportional to the positional error (Eqn. 2) since this pressure difference, and consequently the piston motion, would tend to zero as the desired position was approached.

$$p = \alpha(x_i - x_o) \tag{2}$$

Response characteristics of this system were then obtained by applying a step input to Eqn. 3 and then calculating the resulting effect (Eqn. 4 and 5), which in this case appeared to be a constant amplitude sinusoidal oscillation (Eqn. 6).

$$m\frac{d^2x_0}{dt^2} = \alpha A(x_i - x_o) - F_f \tag{3}$$

$$X_o(s) = \frac{C_1}{s(s+j\omega)(s-j\omega)} \tag{4}$$

$$\text{where} \quad \omega = \left(\frac{\alpha A}{m}\right)^{\frac{1}{2}} \quad \text{and} \quad C_1 = \frac{\alpha A - F_f}{m} \tag{5}$$

$$x_o(t) = \left(1 - \frac{F_f}{\alpha A}\right)\left(1 - \cos(\omega t)\right) \tag{6}$$

However, it should be noted that since the friction term (F_f) was a measure of Coulomb friction the sign of the friction term changed each time the velocity direction changed thereby reducing the magnitude of the oscillation amplitude at each half cycle. The resulting response was therefore a continuous series of progressively smaller (in amplitude) half sine

waves, which oscillated about the desired positional value; with the half waves being joined at points where their first derivatives equaled zero.

The resulting control model for such a system, which employed pistons without return springs, would therefore possess a structure similar that shown in Fig. 6.

4.1.2. System using a piston with a Return Spring

In the case of the conventional control scheme that employed a piston with a spring return it was not possible to close the control loop by simply making the pressure difference between the pistons proportional to the positional error, as in the case without a spring return, since the spring would have caused the system to equilibrate at a position where the pressure force (which would have been proportional to the error) would have equaled the spring force and not at the position where the error would have been zero. This problem was, however, overcome by making the rate of change of pressure ($\dot{p}$), and not the pressure difference (p), proportional to the positional error (Eqn. 7).

$$\frac{dp}{dt} = \alpha(x_i - x_o) \tag{7}$$

Response characteristics for this system were then calculated, as in the previous case, by applying a step input to the control equation (Eqn. 9), which was obtained by differentiating the equation that described the motion of the piston (Eqn. 8), with respect to time (Eqn. 9), and combining it with the loop closure equation (Eqn. 7), and then solving (Eqn. 11) this control equation for the output position value (x_o). It should be noted that despite the fact that a damping term (i.e. the friction term) was not present in the control equation (Eqn. 10), due to the differentiation of Eqn. 8, damping information was reintroduced into the analysis *via* the boundary conditions thereby producing the exponentially decaying oscillation in the final solution (Eqn. 12).

$$m\frac{d^2x_0}{dt^2} + kx_o = pA - F_f \tag{8}$$

$$m\frac{d^3x_0}{dt^3} + k\frac{dx_o}{dt} = A\frac{dp}{dt} \tag{9}$$

$$m\frac{d^3x_0}{dt^3}+k\frac{dx_o}{dt} = \alpha A(x_i-x_o) \tag{10}$$

$$X_o(s) = \frac{C_2(s+C_3)(s+C_4+jC_5)(s+C_4-jC_5)}{s(s+C_6)(s+C_7+jC_8)(s+C_7-jC_8)} \tag{11}$$

$$x_o(t) = c_1+c_2e^{-c_3t}-c_4e^{-c_5t}\sin(c_6t+c_7) \tag{12}$$

The resulting control model for this system, which employed spring return pistons, would therefore possess a structure similar to that shown in Fig. 7.

4.2. Bang-bang Type of Control

Unlike conventional control systems, which maintains continuous control over a system by feeding information about the system's output directly back into the system's input (Fig. 5), bang-bang type of control schemes run systems open loop (Fig. 8) for short periods of time at the end of which changes are made in the adjustable parameters, in order to correct for any errors in the output, before once again allowing the systems to run open loop for a further short period of time, at which point the process would then be repeated until the desired output conditions were reached.

By then selecting the appropriate correction method to be applied to the adjustable parameters during the interval between each open loop cycle it is then possible to tailor such control schemes to any particular control situation. In this specific case the correction method that was chosen involved allowing the length of the next open loop cycle to be directly proportional to the positional error (Eqn. 13). Consequently, as the positional error approached zero the impulse applied to the system also approached zero thereby causing the system to equilibrate at the desired position.

$$\Delta t = \alpha(x_i-x_o) \tag{13}$$

The cycle time was therefore the variable parameter that was changed between open loop cycles in this case, but, if additional control over the system had been required the pressure or rate of change of pressure, could also have been changed between cycles.

In order to obtain response characteristics for the bang-bang control schemes (associated with the systems that did and did not employ return springs) it was only necessary to apply a step input to the equations that described the particular systems (i.e Eqn. 8 and Eqn. 1 for the cases with and without return springs, respectively) and then to solve these equations for the output position (x_o).

4.2.1. System using a Piston without a Return Spring

In the case without the return spring the resulting open loop response to the unit step (Eqn. 14) was found to be quadratic in nature, thus indicating that the overall behaviour of this system, under the control of this bang-bang scheme, would consist of a series of quadratic curves that formed a continuous curve which was also continuous in the first derivative.

$$x(t) = \left(\left(\frac{pA - F_f}{2m}\right)t + v\right)t \tag{14}$$

It should also be noted, however, that without the changes in the variable parameter between the open loop cycles this system would be unstable.

4.2.2. System using a Piston with a Return Spring

In the case of the system that employed the pistons with return springs the bang-bang control scheme produced an output response that was essentially linear, but, with a sine wave superimposed on the underlying curve (Eqn. 15). The overall behaviour of the system was therefore also linear with minor oscillations which only became significant at the later stages of the response.

$$x_o(t) = c_1 + c_2 t - c_3 \sin(c_4 t + c_5) \tag{15}$$

However, unlike the bang-bang case without the return springs this system would remain stable even if no variable parameter changes were performed between open loop cycles since the spring would limit the amount of piston travel by ultimately balancing the constant pressure force.

4.3. Control Scheme Comparisons

On comparing the conventional and bang-bang type of control schemes, as applied to the systems with and without return springs it was found that the faster responses, to a unit step, were obtained from the systems that did not use a return spring (Fig. 9 and 11). However, the lack of a return spring also resulted, in most cases, in a positional error that did not reduce with time, due to the nature of the Coulomb friction, in that once the force induced by the positional error (i.e. $\alpha A(x_i - x_o)$) fell below the static friction value no further motion was possible. Conversely, in the cases with the return spring it was possible to design a system that would converge to the desired position (Fig. 10 and 12). From these observations it was therefore apparent that, in order to ensure that the finger would be able to reach all points in the work space, it was essential to employ pistons that possessed return springs.

This type of system, which employed spring return pistons, was then, therefore, used to allow a direct comparison to be made between the conventional and bang-bang control schemes by comparing the output responses that were produced when each of the control schemes were used to control the behaviour of the system, after a step input had been applied to the system. As a result of this examination it was found that both the conventional (Fig. 11) and the bang-bang (Fig. 12) control schemes produced a comparable response time. However, the character of the response associated with the bang-bang scheme proved to be superior to that of the conventional scheme in that the response associated with bang-bang scheme was approximately linear (Fig. 12) and would therefore result in a constant velocity piston motion (except for a short period near the end of the piston travel); the constant velocity piston motion being advantageous in that it would simplify the design of any subsequent higher level control system.

The bang-bang scheme also possessed the potential to overcome the delay problems that are normally associated with a pneumatic control system, such as settling times and velocity-time delays that exist between the control and the controlled elements (i.e. the valves and the pistons, respectively). In a conventional control scheme these delays would cause the system's response to lag significantly behind the input, due to the existence of the continuous feedback loop, and would ultimately cause the system to become unstable, when these lags began to approach 180°. However, in the case of the bang-bang system the system's stability could be maintained by delaying the response to any new input changes until the system had had time to respond to the prior input changes, provided that the input changes were not too rapid.

It should also be noted that the complexity of the control system would be minimised by adopting the piston actuation scheme described in the technical report by Walker (1985) since this scheme would provide a constant pressure air supply irrespective of the pistons motion (i.e. the air in this case would act as a spring with a spring constant of zero), due to the fact that the pressure source would be the static air pressure present in a steady flow air line: alternatively in a case where the air to the piston would be supplied *via* a valve any motion of the piston would cause a compression or expansion of the air in the piston thereby, in effect, adding an additional spring to the control system.

4.4. Control Equipment

The equipment required to perform the bang-bang control scheme essentially consists of a controller and several peripheral sensors and actuators (fig. 13). The controller itself is designed as a Multibus 1 system with a 68020 processor board, an a-to-d board (to acquire the data from the sensors), a d-to-a board (to control; the motion of the actuators) and an ethernet board (to provide a fast communication link between the controller and the host computer, which would perform the higher level control, such as path planning etc.).

The system's software will then be initially written in C (with, at a later stage, some of the code being written in Assembler) and executed using a real time Unix operating system.

5. Conclusions

The initial study of the finger control has shown that, in terms of system response time and accuracy, a bang-bang type of control scheme would be the preferable way in which to control the type of robot hand described by Walker (1985). It was also noted that such a control scheme might also provide a solution to many of the problems that plague pneumatic control systems, thereby, allowing inexpensive and readily available pneumatic actuators to be used in the design of future robot "hands".

6. References

Patterson, R.W. and Nevill, G.E., "Performance of an Induced Vibration Touch Sensor", SME Technical Paper No. MS85-575, June 1985.

Walker, G., "An Initial Finger Design for an Industrial Hand: Part I", CIMS Technical Report No. 196, December 1985.

Walker, G., "A Proposed Slip and Texture Sensor", CIMS Technical Report No. 207, March 1986.

7. Nomenclature

A - piston area

c - constant in the time domain

C - constant in the s domain

e - error

F_f - friction force

j - $\sqrt{-1}$

k - spring constant

m - mass being displaced

p - pressure difference between pistons

s - independent variable in the s domain

t - time

v - piston velocity at the end of an open loop cycle

x_i - input displacement in the time domain

x_o - output displacement in the time domain

$x0$ - initial displacement

$x1$ - final displacement

X_o - output displacement in the s domain

α - feedback gain

β - feedback gain

θ_i - input angular displacement of finger joint

θ_o - output angular displacement of finger joint

ϕ_i - input degree of valve closure

ϕ_o - output degree of valve closure

ω - angular frequency

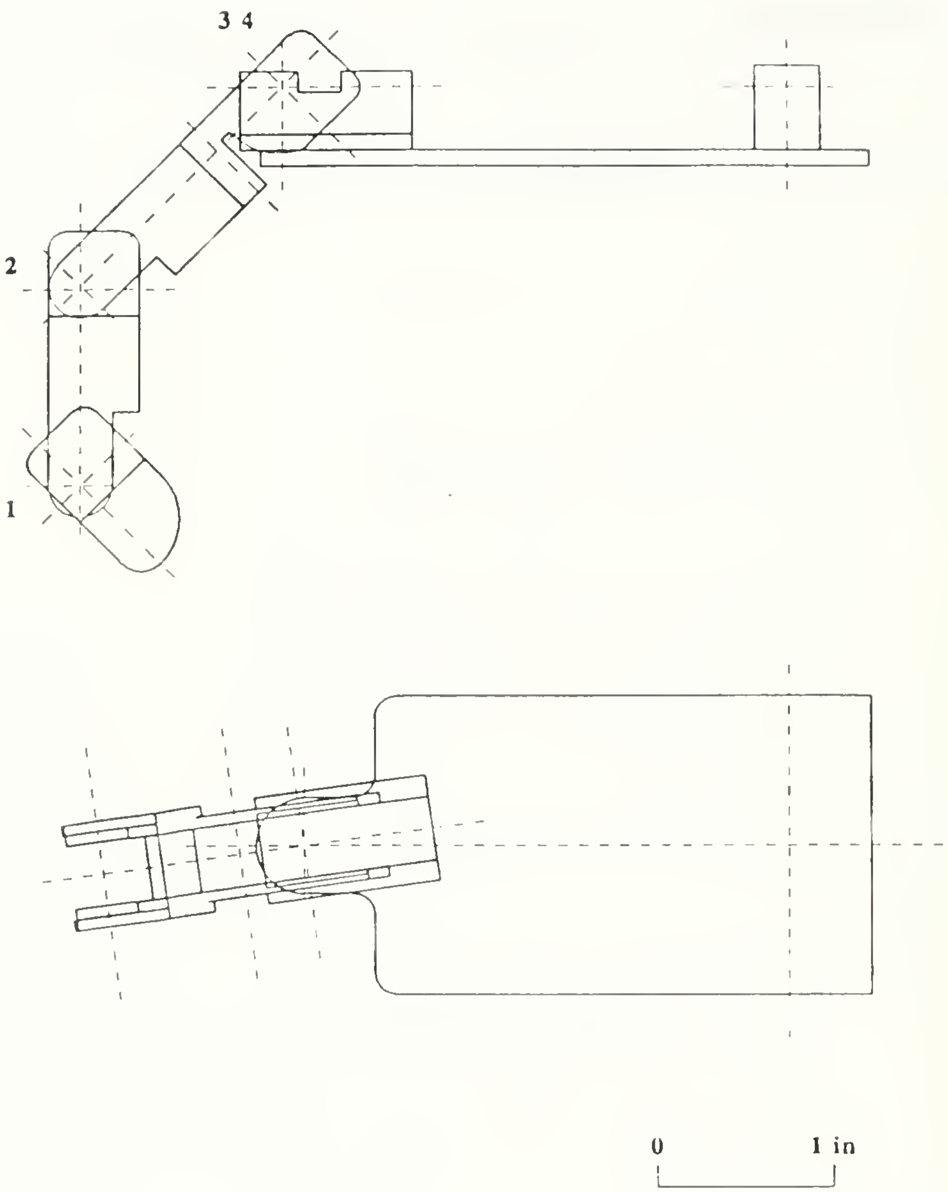

Fig. 1 A schematic drawing of the new finger design

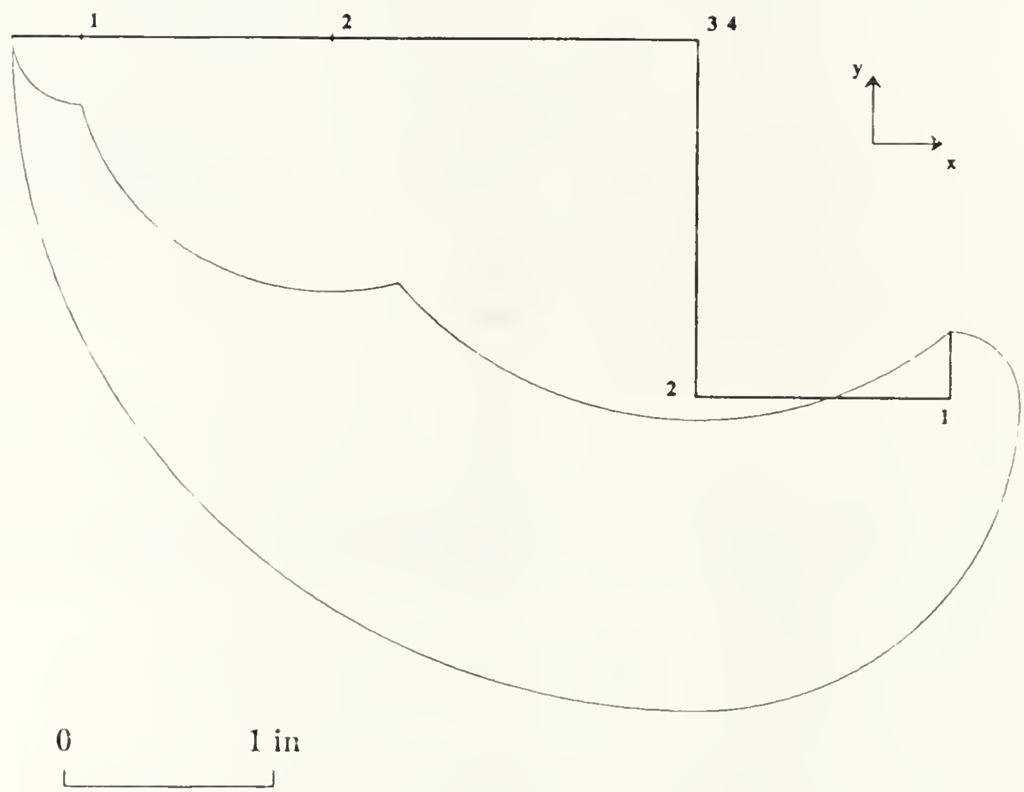

Fig. 2 The working area of the new finger design

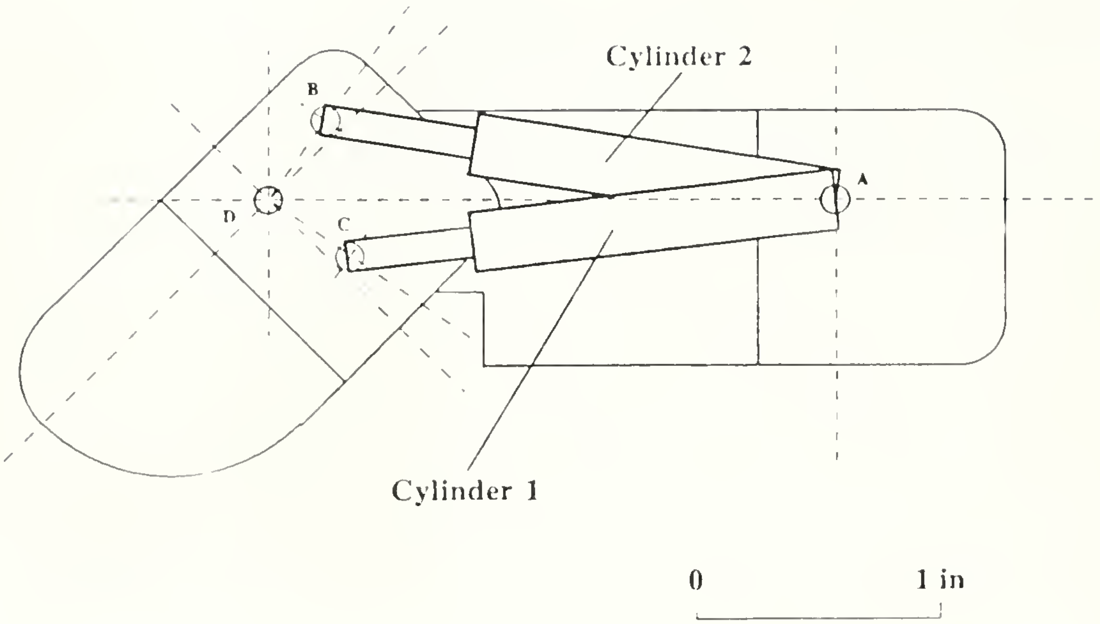

Fig. 3 A schematic drawing of the new actuation scheme

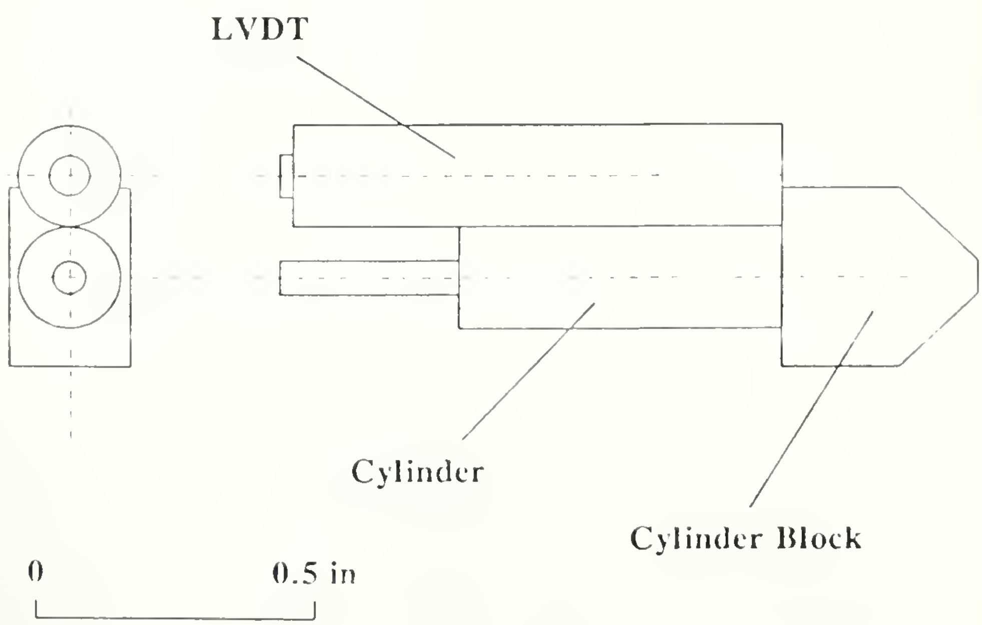

Fig. 4 A schematic drawing of the cylinder-LVDT arrangement

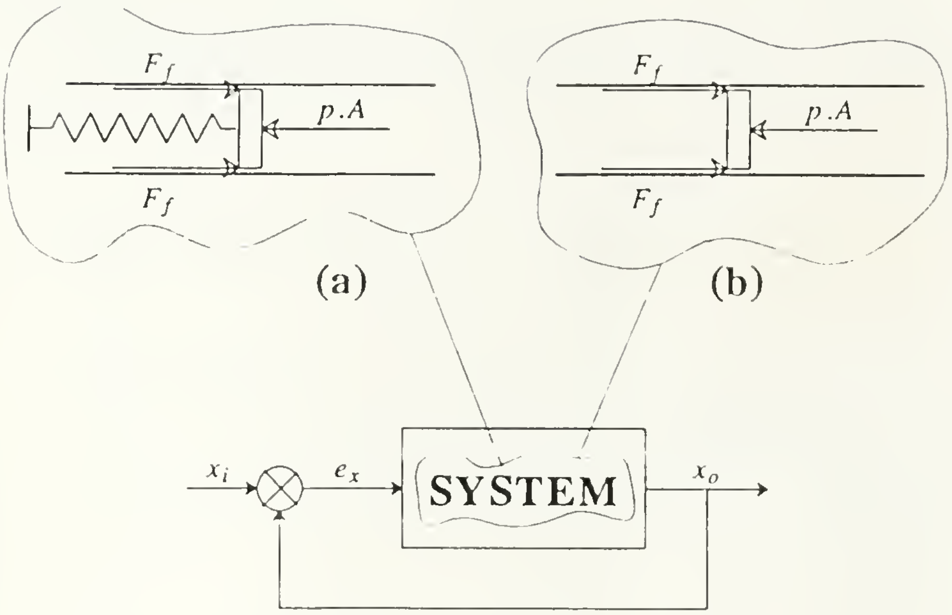

Fig. 5 An example of a conventional control scheme for a system (a) with and (b) without a return spring

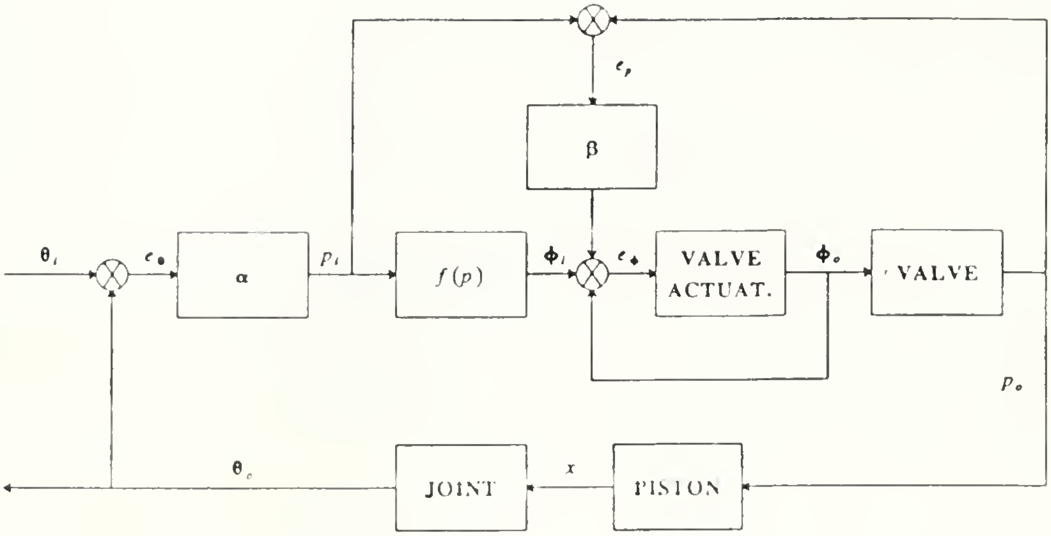

Fig. 6 The control model for a system that employs pistons without return springs

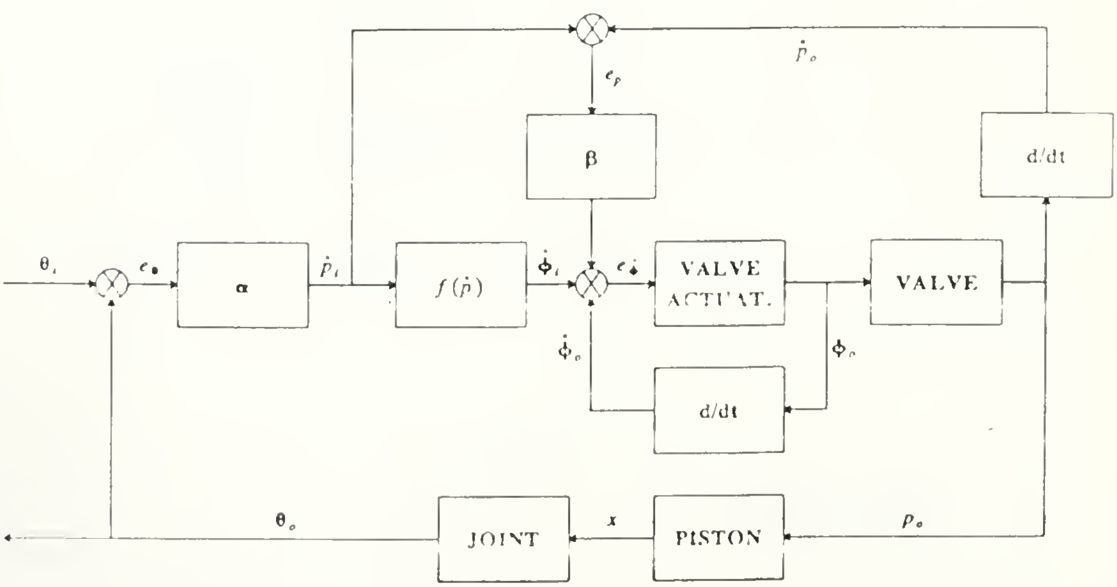

Fig. 7 The control model for a system that employs pistons with return springs

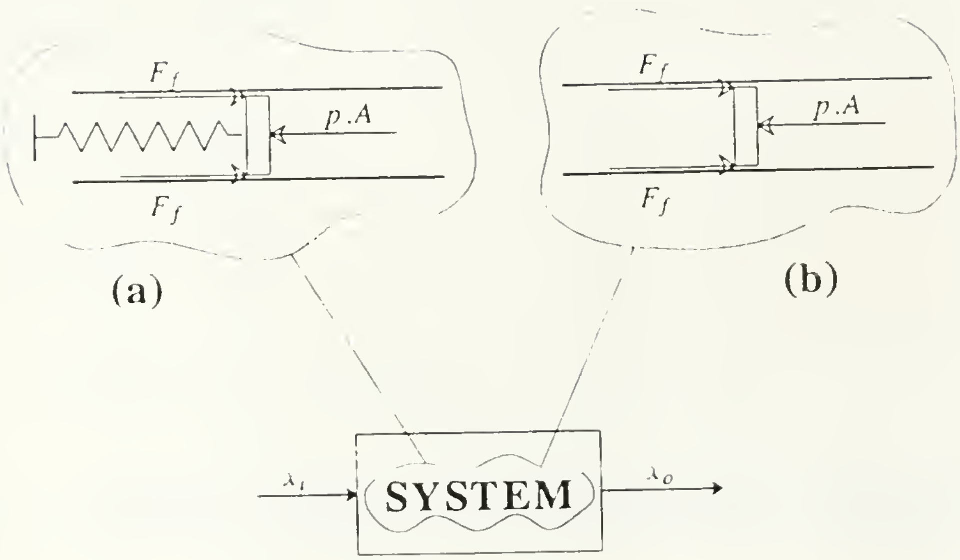

Fig. 8 An example of a bang-bang type of control scheme for a system (a) with and (b) without a return spring

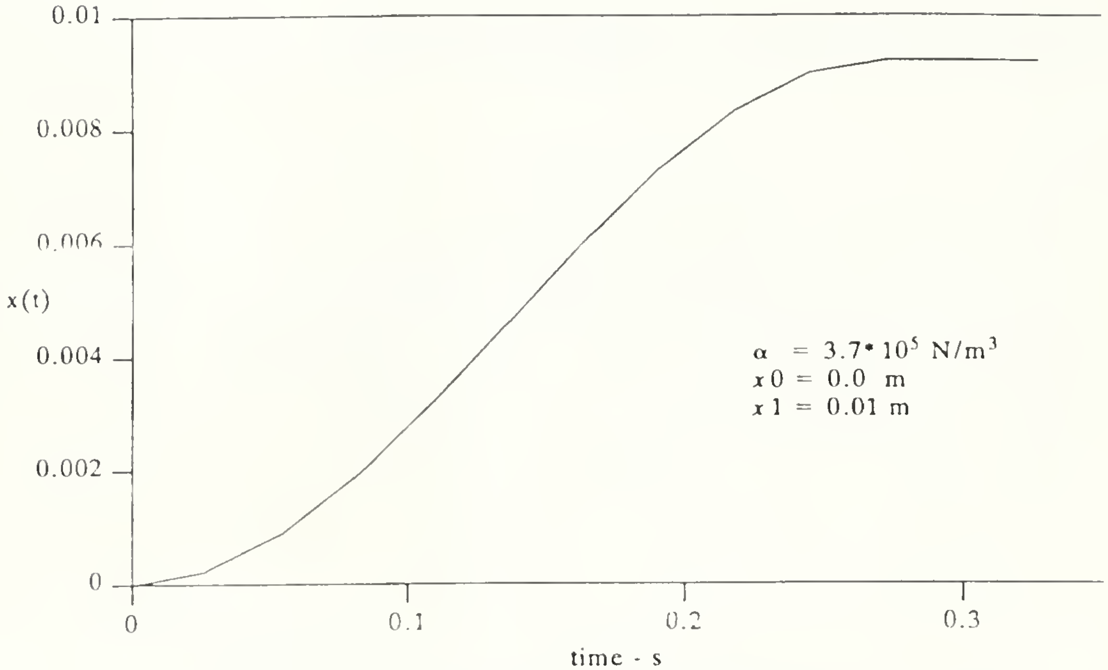

Fig. 9 The response of a cylinder without a spring, and which is controlled by a conventional control loop, to a unit step

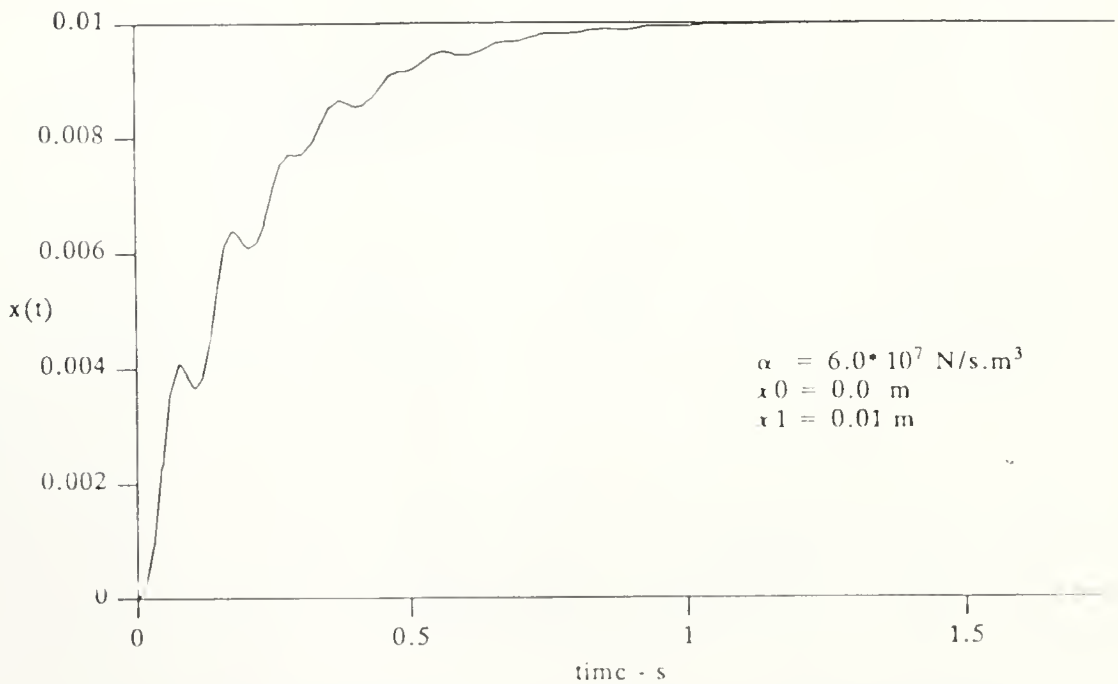

Fig. 10 The response of a cylinder with a spring, and which is controlled by a conventional control loop, to a unit step

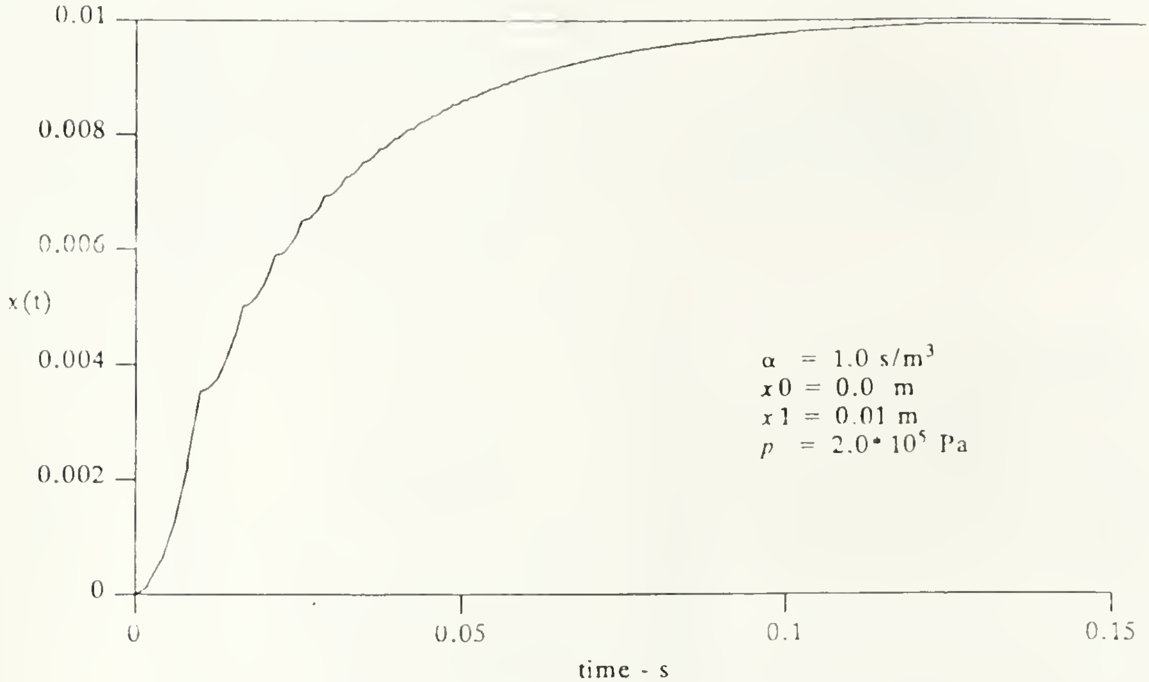

Fig. 11 The response of a cylinder without a spring, and which is controlled by a bang-bang control scheme, to a unit step

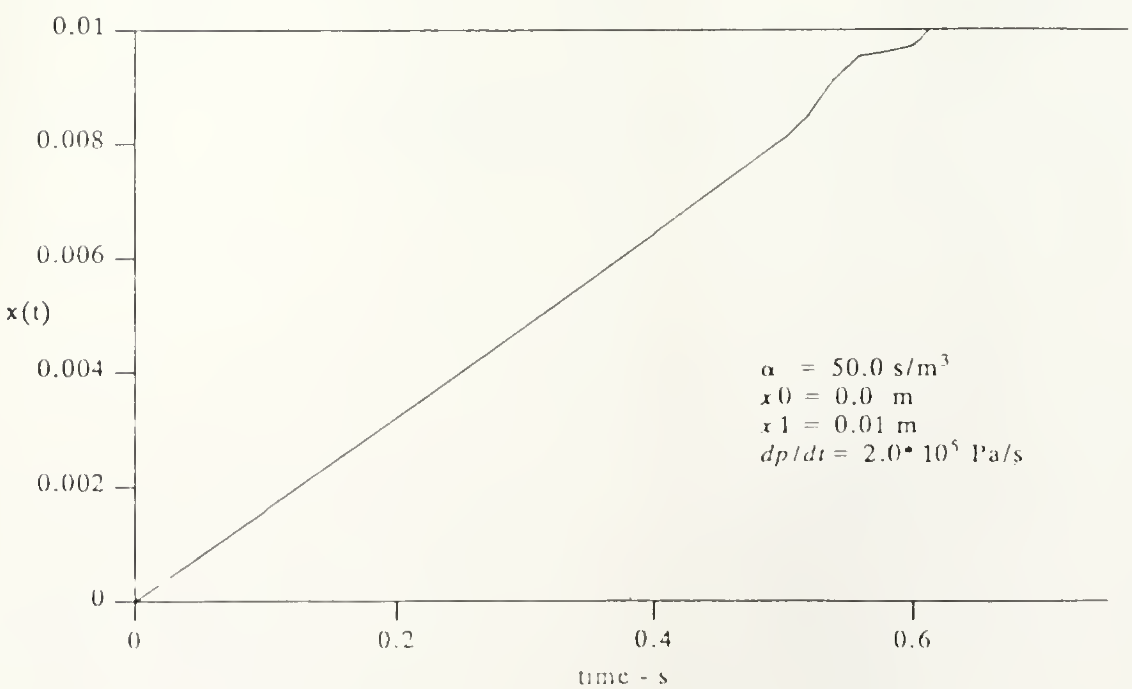

Fig. 12 The response of a cylinder with a spring, and which is controlled by a bang-bang control scheme, to a unit step

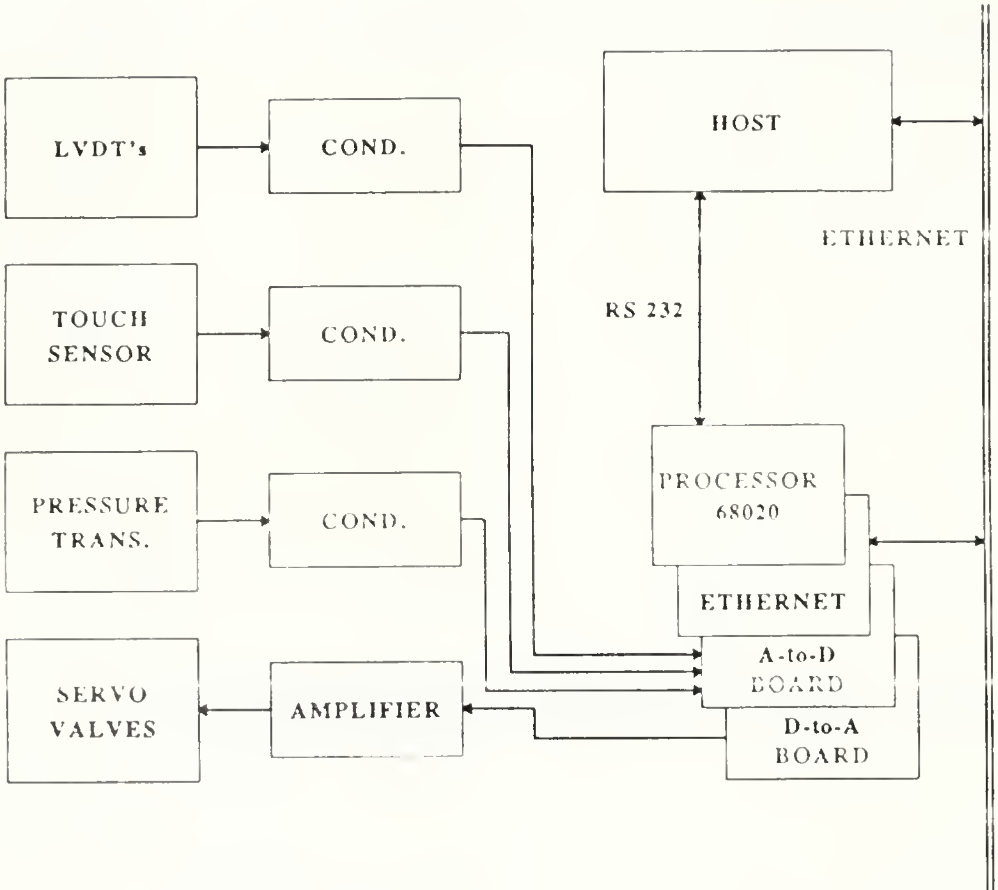

Fig. 13 A schematic diagram of the control hardware

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